*Journal of Engineering Physics and Thermophysics, Vol. 81, No. 1, 2008*

# EVAPORATIVE COOLING OF WATER IN COMPLEX-CONFIGURATION FILM SPRAY ZONES

**A. I. Petruchik, A. D. Solodukhin,**
**and S. P. Fisenko**

UDC 536.42:661.426

*Evaporative cooling of water in film flows in a double-corrugation spray zone is investigated. It is shown that the main contribution to the increase in the efficiency of evaporative cooling in such spray zones is made by artificial turbulization of air flow on the corrugations of the guards. For a sprinkler with guards of arbitrary configuration a method of engineering calculation of the efficiency of evaporative cooling is suggested.*

**Introduction.** Evaporative cooling of water in cooling towers is widely used in industry and power engineering. Until recently asbestos-cement spray zones made in the form of plane vertical guards were commonly used in cooling towers. At the present time, in order to raise the efficiency of evaporative cooling of water in cooling towers plastic guards of complex configuration are beginning to be used for spray zones that are noticeably more advantageous as concerned their thermal efficiency. The designers of new spray zones are facing the problem of rapid and rather accurate assessment of their thermal efficiency on the basis of reliable physical methods of the heat- and mass transfer theory without employing expensive test rigs. In addition to the thermal efficiency, in selecting spray zones attention is also to be paid to such problems as the cost, longevity, convenience of mounting, fire safety, frost resistance, as well as stability with respect to the chemical composition of circulating water.

In the present work we suggest a mathematical model of evaporative cooling of water films in spray zones of complex configuration, specifically, in the spray zones made from polyvinyl chloride sheets of double corrugation (Fig. 1). Our earlier mathematical model of evaporative cooling of water films falling down two adjoining plane guards [1] was used as the basis of our new mathematical model.

As shown in [1], the mass density of irrigation $q_{\rm w}$ per unit length of the spray zone is an important parameter that influences the efficiency of sprinkler operation. In contemporary cooling towers $q_{\rm w} = 0.02–0.03$ kg/(m·sec), with the average thickness of the water film over the spray zone being of about 0.2 mm, and the average velocity of water film falling $\langle v_{\rm w}\rangle \approx 0.1$ m/sec. The Reynolds number for the film flow of water is $\mathrm{Re_w} \approx 40$, which corresponds to a laminar mode of flow with formation of long gravitational waves of length of about 2 cm [2].

Following [3], in the mode of laminar gravitational flow the thickness $h$ of the liquid film falling down the plate deflected from the vertical by an angle $\beta$ is defines as

$$h = \left(\frac{3}{4}\frac{\nu_{\rm w}}{g}\frac{4q_{\rm w}}{\rho_{\rm w}\cos\beta}\right)^{1/3} . \tag{1}$$

Over the stabilized portion of flow, where the gravity force is balanced out by viscous forces, the section-average velocity of falling film $\langle v\rangle$ depends on the film thickness as follows:

$$\langle v_{\rm w}\rangle = \frac{gh^2\cos\beta}{3\nu_{\rm w}} . \tag{2}$$

The second important parameter that influences the evaporative cooling of water films in the spray zones is the mass flow rate of the air that passes between two adjoining guards. Its average velocity is usually equal to about

---

A. V. Luikov Heat and Mass Transfer Institute, National Academy of Sciences of Belarus, 15 P. Brovka Str., Minsk, 220072, Belarus; email: fsp@hmti.ac.by. Translated from Inzhenerno-Fizicheskii Zhurnal, Vol. 81, No. 1, pp. 171–175, January–February, 2007. Original article submitted March 1, 2007.

1062-0125/08/8101-0182©2008 Springer Science+Business Media, Inc.

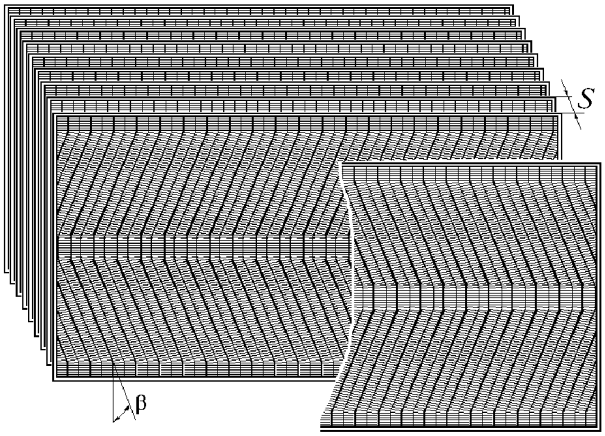

Fig. 1. A block of film spray zone consisting of polyvinyl chloride sheets of double corrugation.

1 m/sec, with the Reynolds number $\mathrm{Re}_{\mathrm{a.w}}$ determined in terms of the transverse distance between the guards being equal to about $4{\cdot}10^3$. At such Reynolds numbers, the flow, by virtue of its hydrodynamic instability, becomes turbulent if the channel length is sufficient. We note that for contemporary cooling towers the mass flow rate of water is approximately equal to the mass flow rate of the air that passes through the spray zones. It has been established that the efficiency of evaporative cooling depends on the following parameters: temperature and humidity of the air entering the spray zone, the initial temperature of water, and the shape of the guards [4, 5].

**Mathematical Model.** In order to model evaporative cooling of water films in spray zones of complex shape two new parameters were introduced by us into our earlier model. Firstly, for the water films that fall down corrugated guards we take into account the actual increase in the length of a guard as well as the influence of corrugations on the film thickness $h$ and on the average velocity of the fall of films $\langle v_{\mathrm{w}} \rangle$. Secondly, to take into account the turbulent character of air flow between two guards with double corrugation new expressions were introduced for the heat- and mass transfer coefficients.

Just as before [1, 4, 5], the mathematical model represents a boundary-value problem for a system of ordinary differential equations that describe a change in the parameters of the film and vapor-air mixture with the downward coordinate $z$.

The first of the equations of the mathematical model describes the change in the film thickness $h(z)$ due to water evaporation:

$$\frac{dh(z)}{dz} = -\frac{\gamma(\mathrm{Re}_{\mathrm{a.w}})(\rho_{\mathrm{s.v}}(T_{\mathrm{w}}(z)) - \rho_{\mathrm{v}}(z))}{\rho_{\mathrm{w}} \langle v_{\mathrm{w}} \rangle}, \tag{3}$$

the second equation characterizes the change in the film section-average temperature of water $T_{\mathrm{w}}(z)$ due to contact with a cold air and evaporation:

$$\frac{dT_{\mathrm{w}}(z)}{dz} = \frac{\alpha(\mathrm{Re}_{\mathrm{a.w}})(T_{\mathrm{a}}(z) - T_{\mathrm{w}}(z)) - r\gamma(\mathrm{Re}_{\mathrm{a.w}})(\rho_{\mathrm{s.v}}(T_{\mathrm{w}}(z)) - \rho_{\mathrm{v}}(z))}{c_{\mathrm{w}}\rho_{\mathrm{w}} h \langle v_{\mathrm{w}} \rangle}, \tag{4}$$

the third equation describes the change in temperature of the vapor-air mixture $T_{\mathrm{a}}(z)$ averaged over the spray zone cross section:

$$\frac{dT_{\mathrm{a}}(z)}{dz} = -\frac{2\alpha(\mathrm{Re}_{\mathrm{a.w}})(T_{\mathrm{w}}(z) - T_{\mathrm{a}}(z))}{\langle v_{\mathrm{a}} \rangle S \rho_{\mathrm{a}} c_{\mathrm{a}}}, \tag{5}$$

TABLE 1. Temperature of the Cooled Water at a Fixed Difference in the Water Temperature of 10.9°C, with a Density of Irrigation of 5 $m^3/(m^2 \cdot h)$ (according to B. L. Sverdlin's experimental data)

| Spray zone tier | Height, m | Temperature of heated water, °C | Air velocity, m/sec | Temperature of cooled water, °C | |
|---|---|---|---|---|---|
| | | | | experiment | calculation |
| 1 | 0.7 | 42.90 | 1.01 | 32.01 | 32.0 |
| 1,5 | 1.1 | 40.17 | 0.98 | 29.28 | 29.2 |
| 2 | 1.4 | 38.94 | 0.96 | 28.05 | 27.8 |

and, finally, the fourth equation of the mathematical model allows one to calculate the change in density of water vapors $\rho_v(z)$ in the vapor-air mixture averaged over the sprinkler cross section:

$$\frac{d\rho_v(z)}{dz} = -\frac{2\gamma(\mathrm{Re}_{a.w})(\rho_{s.w}(T_w(z)) - \rho_v(z))}{\langle v_a \rangle S} . \tag{6}$$

For a turbulent flow past double-corrugation guards we used the available experimental data (see Table 1) and found a new numerical factor in the expression for the heat transfer coefficient:

$$\alpha = \frac{0.05\mathrm{Re}_{a.w}^{0.8}\lambda_a}{H - z} . \tag{7}$$

It should be emphasized that the heat transfer coefficient given by Eq. (7) is higher than that for a turbulent flow past a flat plate [6]. With allowance for the geometry of the problem, the Reynolds number for the air flow past a water film is defined as

$$\mathrm{Re}_{a.w} = \frac{(H - z)\rho_a(\langle |v_a| \rangle + \langle |v_w| \rangle)}{\mu_a} , \tag{8}$$

where $\langle |v_a| \rangle + \langle |v_w| \rangle$ is the relative velocity of the opposite air flow past the falling water film.

Using the analogy between the heat and mass transfer [7], the mass transfer coefficient is defined as

$$\gamma = \frac{0.05\mathrm{Re}_{a.w}^{0.8}D}{H - z} . \tag{9}$$

For a turbulent flow past a flat plate [6] there are the well-known dependences of the heat- and mass transfer coefficients:

$$\alpha = \frac{0.025\mathrm{Re}_{a.w}^{0.8}\lambda_a}{H - z} . \tag{10}$$

$$\gamma = \frac{0.025\mathrm{Re}_{a.w}^{0.8}D}{H - z} . \tag{11}$$

The heat- and mass transfer coefficients (10) and (11), unlike the coefficients (7) and (9), do not take into account forced small-scale turbulization of the vapor-air flow on the spray zone corrugations and were obtained for a turbulent air flow past a flat plate. We recall that for a laminar flow the heat- and mass transfer coefficients have the form [2, 6]

$$\alpha(\mathrm{Re}_{a.w}) = \frac{0.324\mathrm{Re}_{a.w}^{0.5}\mathrm{Pr}_a^{0.33}\lambda_a}{H - z} , \tag{12}$$

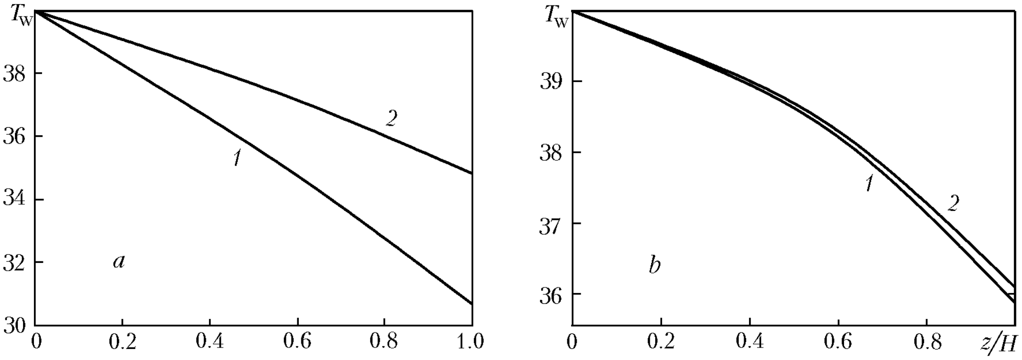

Fig. 2. Profiles of the water film temperature $T_w$ over the height of the guards: a) 1) double-corrugation spray zone with the angle of inclination of the main corrugations to the horizon of $60^o$; the turbulent coefficients of heat and mass transfer were taken from expressions (7) and (9); 2) for plane vertical guards with turbulent air flow past them the coefficients of heat and mass transfer were calculated from (10) and (11); b) 1) for double-corrugation guards the angle of inclination of the main corrugations to the horizon is $60^o$ with the use of the laminar coefficients of heat and mass transfer from expressions (12) and (13); 2) for plane guards.

$$\gamma\,(\mathrm{Re}_{\mathrm{a.w}}) = \frac{0.324\mathrm{Re}_{\mathrm{a.w}}^{0.5}\mathrm{Pr}_{\mathrm{a}}^{0.33}D}{H - z}\,. \tag{13}$$

**Results of Calculations.** In the first numerical experiment the coefficients (7) and (9) were determined on the basis of the available experimental data for a fixed structure of a spray zone ($\beta$ = $30^o$C, $S$ = 40 mm). The results of this calculation given in Table 1 were obtained for the temperature of the air entering a spray zone $T_a$ = $22^o$C and its relative humidity $\varphi$ = 60%. As is seen from the table, the use of the coefficients (7) and (9) for calculation of other experiments ensures coincidence of the results of calculation with experimental data with a high degree of accuracy. In particular, the relative difference does not exceed 2%.

In the second series of numerical experiments the degree of influence exerted by the heat- and mass transfer coefficients as well as by the main geometrical parameters of a guard was investigated. For this purpose, four water temperature profiles along the guard were calculated with the aid of different heat- and mass transfer coefficients. The dependences of the water film temperature on the dimensionless coordinate along the guard are shown in Fig. 2. They were obtained at $T_{w0}$ = $40^o$C, $T_{a0}$ = $22^o$C, and $\varphi$ = 60%; the specific water flow rate per unit length of a guard was 0.02 kg/(m·sec), the velocity of the ascending air was 1 m/sec, the guard height was 0.7 m, the distance between the adjoining guards of the spray zone was 40 mm.

The curves in Fig. 2 were obtained for the same specific density of irrigation (per unit length). As is seen from the calculation results given, the determining factor of increasing the efficiency is vapor-air flow turbulization. In calculations with turbulent coefficients of heat and mass transfer even for plane guards the depth of water cooling in the spray zone increases substantially as compared to the laminar solution. Moreover, the velocity of water film fall down a corrugated guard is lower than that over a vertical plate.

We note that the thermal efficiency of spray zones $\eta$ is calculated as follows [8]:

$$\eta = \frac{T_{\mathrm{in}} - T_{\mathrm{f}}}{T_{\mathrm{in}} - T_{\mathrm{lim}}}\,,$$

where $T_{\mathrm{in}}$ and $T_{\mathrm{f}}$ are the temperature of water at the inlet and exit from a spray zone; the limiting temperature $T_{\mathrm{lim}}$ is equal to the wet-bulb temperature. The integral parameter of the operation of a spray zone $\eta$ is independent of the humidity of the entering air and for the data given in Fig. 2a it is equal to 0.35 (curve 1) and 0.19 (curve 2), respectively, whereas in Fig. 2b $\eta = 0.15$ (curve 1) and $\eta = 0.13$ (curve 2).

The higher the thermal efficiency of a spray zone, the higher the relative humidity and temperature of the vapor-air mixture flowing out of a spray zone, with the mass flow rate of the evaporated water being equal to about 1% of the mass flow rate of the water flowing into the spray zone, and it increases proportionally to the increase in the thermal efficiency of the spray zone.

**Conclusions.** A mathematical model and a computer program to calculate evaporative cooling of water films falling down the guards of a spray zone of complex configuration have been developed. As the basis we used our previous mathematical model of evaporative cooling of laminar water films on plane vertical guards [1]. In order to take into account the geometry of double-corrugation guards and the character of water and air flow, additional parameters and new numerical factors were introduced into the expressions for the heat- and mass transfer coefficients (7)–(9). It is shown that for all types of spray zones the cooling is nonuniform over the length — it mainly occurs in the lower third part of the spray zone, where a relatively cool air stream unsaturated with moisture flows past the phase interface of the water film.

A comparison of the results of calculation for guards of different geometries and different patterns of cooling air flow past the water film has made it possible to reveal the main factor that influences the intensification of evaporative cooling in spray zones of contemporary design. This is artificial turbulization of the vapor-air allowing one to intensify the removal of heat and vapor from the phase interface.

Two-dimensional simulation of evaporative cooling of water on plane guards with a laminar mode of flow has also shown that the limiting process is the removal of heat and water vapors from the phase interface [8]. Artificial turbulization of the vapor-air stream that originates rapidly during flow past the periodic structure of corrugation in a spray zone makes it possible to effectively remove heat and vapor from the film. Naturally, this leads to an increase in the hydraulic resistance of the spray zone [7]. It seems that in the upper part of the guards one can dispense of corrugation without a substantial loss in the thermal efficiency of the spray zone, but this will reduce its hydraulic resistance.

It should be noted that the use of new corrugated spray zones in evaporative cooling towers will not yield an appreciable gain, as is shown in Fig. 2a. The reason is that the increased hydraulic resistance of new corrugated spray zones will cause a decrease in the average velocity of the ascending vapor-air flow and in the Reynolds number and thus will reduce their thermal efficiency [5].

The developed mathematical model makes it possible to carry out forecasting calculations of the efficiency of new film spray zones of complex shape. For evaluating calculations of new constructions of tower guards we can recommend to use the mathematical model of [4, 5] in which such parameters are used as the specific density of irrigation, the average velocity of the ascending air, the length of the guards, and the distance between the adjoining guards. As expressions for the heat- and mass transfer coefficients it is advisable to use Eqs. (7)–(9). A comparison of this calculation with the data obtained with the aid of the heat- and mass transfer coefficients (12), (13) makes it possible to rather accurately evaluate the increase in the efficiency of evaporative cooling on the guards of a tower.

We note that investigation of nonstationary effects in evaporative cooling of water films on plane guards of a tower was made in [9].

## NOTATION

$c$, specific heat, J/(kg·K); $D$, diffusion coefficient, $m^2$/sec; $g$, free fall acceleration, m/$sec^2$; $H$, height of a tower, m; $h$, liquid film thickness, m; Pr, Prandtl number; $q$, mass density of irrigation, kg/(m·sec); $r$, specific latent heat of vaporization, J/kg; Re, Reynolds number; $S$, distance between guards, m; $T$, temperature, $^{o}$C; $v$, velocity of water film falling, m/sec; $z$, vertical coordinate, m; $\alpha$, heat transfer coefficient, W/($m^2$·$^{o}$C); $\beta$, angle of deflection of main corrugations from the vertical, deg; $\gamma$, mass transfer coefficient, m/sec; $\Delta$, difference in values; $\eta$, thermal efficiency; $\lambda$, thermal conductivity, W/(m·$^{o}$C); $\mu$, dynamic viscosity, kg/(m·sec); $\nu$, kinematic viscosity, $m^2$/sec; $\rho$, density, kg/$m^3$; $\varphi$, relative humidity of air, %. Subscripts: a, air; a.w, air flow past water; in, value at the inlet; f, final value; lim, limiting value; s.v, saturated vapor; v, vapor; w, water; 0, initial value.

## REFERENCES

1. A. I. Petruchik and S. P. Fisenko, Mathematical modeling of evaporative cooling of water films in water-cooling towers, *Inzh.-Fiz. Zh.*, **72**, No. 1, 43–49 (1999).
2. V. M. Olevskii (Ed.), *Film Heat- and Mass Exchange Equipment* [in Russian], Khimiya, Moscow (1988).
3. G. Gimbutis, *Heat Transfer in Gravitational Liquid Film Flow* [in Russian], Mokslas publishers, Vilnius (1988).
4. A. I. Petruchik, A. D. Solodukhin, and S. P. Fisenko, Simulation of cooling of water droplet and film flows in large nature wet cooling towers, *Inzh.-Fiz. Zh.*, **74**, No. 1, 45–49 (2001).
5. S. P. Fisenko, A. I. Petruchik, and A. D. Solodukhin, Evaporative cooling of water in a natural draft cooling tower, *Int. J. Heat Mass Transfer*, **45**, No. 23, 4683–4694 (2002).
6. A. A. Zhukauskas, *Convective Transfer in Heat Exchangers* [in Russian], Énergiya, Moscow (1982).
7. G. F. Hewitt, G. L. Shires, and Y. V. Polezhaev (Eds.), *International Encyclopedia of Heat & Mass Transfer*, CRC Press, Boca Raton (1997).
8. A. I. Petruchik and S. P. Fisenko, Two-dimensional calculation of the parameters of a steam-air mixture in a film-type heat and mass exchanger, *Inzh.-Fiz. Zh.*, **76**, No. 5, 81–86 (2003).
9. G. V. Dashkov, E. F. Nogotov, N. V. Palvyukevich, and V. D. Tyutyuma, Nonstationary heat and mass transfer in evaporative cooling of flowing liquid films, *Inzh.-Fiz. Zh.*, **79**, No. 1, 3–10 (2006).